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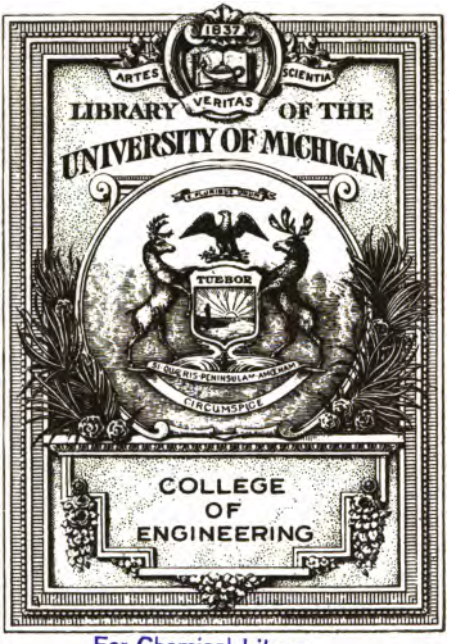

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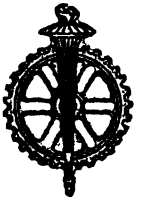
STEAM PIPING

ITS ECONOMICAL DESIGN AND CORRECT LAYOUT

BY

A. LANGSTAFF JOHNSTON, JR.



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PREFACE

Mr. Johnston believes, as a deduction from his own professional observation and experience, that in the majority of industrial plants the steam-piping system has received insufficient attention, and that as a result wastes of both installation investment and operating cost are prevalent.

In this volume, consolidated and revised from a series of articles published in THE ENGINEERING MAGAZINE in 1915, he analyzes the factors governing the flow of steam in pipes, and presents a group of curves for use in solving the problems of practical installation and determining the most economical size of pipe to select for any given set of conditions.

While the first chapter includes in its theoretical discussions all pressures from 0 to 250 pounds, two supplementary chapters take up respectively the special problems of low-pressure systems, and the enormous but too often neglected economies obtainable from the utilization of exhaust steam, and the employment of the plain reciprocating engine, with exhaust heat as a by-product, in place of much more costly equipment sometimes installed.

CHARLES BUXTON GOING

March, 1916.

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CONTENTS

CHAPTER I. HOW TO FIND THE RIGHT PIPE SIZES 1

Defects of "Standard Practice"—General Neglect of the Piping System—Great Savings Possible—Conditions Governing the Flow of Steam in Pipes—Explanation of the "Steam-Velocity Curves" Used in this Volume—How They are Employed—Examples—The Pressure-Drop Curve—Illustration and Discussion of Its Use—Effect of Fittings in Increasing Pressure Drop—Allowances for This Resistance—Heat Loss by Radiation—Curve Showing Most Economical Size of Pipe Under Any Given Combination of Conditions—Examples of Its Use—Piping as Usually Installed is Much too Large—Losses Resulting from this Common Fault of Practice—A Typical Instance Studied by Aid of the Curves—Startling Conclusions.

CHAPTER II. SPECIAL CONDITIONS AFFECTING LOW-PRESSURE SYSTEMS 27

Laying Out the Piping for a Steam Heating Plant—Ample Radiating Surface the First Requisite—The Circulating System Comes Next in Importance—Proper Proportioning of Pipe Sizes will Minimize the Radiation and Friction Losses—Effects of

Increased Steam Flow upon Radiation Loss and Pressure Drop—Coefficient of Condensation—Steam-Velocity Curves and Their Practical Use — Drop - in - Pressure Curves—How They Are Interpreted and Used in Practice—What Pressure Drop is Allowable and Most Economical—Explanation of the Curves, with Examples—Return Mains and Their Proper Sizes—Table of Constants.

CHAPTER III. SAVINGS OBTAINABLE FROM EXHAUST STEAM..... 47

Widespread Ignorance or Neglect of Economical Power-Plant Operation—Reasons for This Indifference in the Past—Present Means for Measuring and Remedying Steam-Plant Losses—Maximum Economy Obtainable from a Steam Turbine—Superior Efficiency of the "By-Product Steam Engine"—A Practical Case Making This Clear—An Over-All Thermal Efficiency of 75 to 90 per cent Obtained from a Simple Reciprocating Engine when the Exhaust is Used for Heating—Curve Showing Value of Heat in Exhaust—Effect of Back Pressure on Engine Efficiency—How to Find the Most Economical Conditions for Plant Operation—Curve Giving the Solution of the Equation Graphically—Conclusions.

STEAM PIPING

Its Economical Design and Correct Layout

CHAPTER I

HOW TO FIND THE RIGHT PIPE SIZES

THE common expression "standard practice," though convenient, has done much to hinder the economical design and development of steam machinery, whereas the science of electrical engineering is too young and its development has been too rapid to be so hampered. The demand created by the wonderful possibilities of the latter science has been met by the steam engineer as well as the electrical engineer, and there seems to be no limit to the size of units these engineers are capable of producing, there being under construction now a 35,000-kilowatt steam turbine, and boilers of 2,365 horse-power rating are in use, operating at 5,500 horse-power each.

The manufacturers of steam machinery are

experimenting continually to improve the efficiency of their products, realizing that a small increase in efficiency, in a large central-station plant, will result in a material yearly operating-cost saving. Fortunately for our natural resources, the industrial power-plant owner, who operates about 80 per cent of the total steam horse-power in the United States, is also beginning to realize that a percentage saved, whether on one dollar or one million dollars, is a saving and worth considering.

The one item in the majority of industrial plants that has received the least engineering attention is the steam piping. Of course, in large central-station plants, where great quantities of steam have to be handled, the piping layout is an important item and receives the engineering consideration due it. But with the average of industrial plants, even those owned by large corporations which carefully design the piping to accompany their products, the size of steam piping used in the factories seems to have been selected according to "standard practice," "rule of thumb," or some such method. There will be found in the average small plant of today the same size of live-steam

piping supplying a 500-kilowatt plant as is used in a modern 5,000-kilowatt plant. I can recall only one large industrial plant that came under my own observation where the piping layout, so far as concerns size, was carefully considered and economically designed.

It is the aim of this book to bring out some of the laws governing the flow of steam in pipes, and the resulting losses; also to present several theoretical curves in convenient form, to be used in designing the most economical size of pipe to meet certain conditions of flow, pipe diameter, and pressure.

The two losses of energy in conveying steam in pipes are: first, that due to radiation, which is governed by the difference in temperature of the steam and the external air, area of pipe surface, and type of insulating covering used; second, that due to pressure drop caused by the friction of the steam against the surface of the pipe, which is governed by the velocity and density of the steam flowing. This loss in pressure varies as the square of the velocity; that is, by doubling the velocity the pressure drop is increased fourfold, and as we are in the habit of speaking of a pipe being worked at a certain ve-

locity, and not a certain pressure loss, a flow-velocity curve should be of interest.

It will be well first to explain the method followed in calculating and drawing the accompanying curves. Marks and Davis steam tables are used. The curves themselves, being much too large for reproduction on the pages of a handbook, are separately printed on suitable sheets which will be found folded and inserted within the back cover of this volume. They may thus be most conveniently spread out separately for reference or study.

Steam-Velocity Curves

Curve No. 1 is a steam-flow pipe-diameter velocity curve drawn for 100-pounds gauge pressure saturated steam and derived from the formula:

$$\text{Velocity} = \frac{\text{Flow of steam} \times \text{Specific volume of steam}}{\text{Area of pipe}}$$

This reduces to

$$V = \frac{F \times S}{25 \times A} \qquad \text{(I)}$$

in which

V = velocity of steam in feet per second
F = flow of steam in pounds per hour

S = specific volume of steam at pressure considered
A = internal area of pipe in square inches

By substituting in this equation, we obtain the velocity of steam at 100-pounds gauge pressure corresponding to various flows and pipe diameters. As the velocity varies directly with the flow, these curves are straight lines and easily drawn and read. From equation (I) it will be seen that the velocity is proportional to the specific volume. Hence to obtain the velocity for any other pressure than the standard at which the curve is drawn, it is necessary only to multiply the velocity at 100-pounds pressure by the ratio of the specific volumes, *i.e.*,

$$\text{Velocity at pressure X or } V_X = \frac{S_X}{S_{100}} \times V_{100} \quad \text{(II)}$$

By this method, the pressure correction curves (Curve No. 2) for the various pressures are obtained, and are, for the same reason stated above, straight-line curves.

To use these curves, (suppose) it is desired to find the velocity of 20,000 pounds of steam per hour flowing in a 6-inch pipe at 75-pounds gauge pressure. Referring to Curve No. 1, read vertically downward from the 20,000-

pound flow line until the 6-inch pipe-diameter curve is intersected, then horizontally until the 75-pound pressure curve in the second quadrant is intersected, then vertically to the velocity abscissa, which shows a velocity of 135 feet per second.

In like manner, the flow corresponding to any given pressure, pipe diameter, and velocity may be found. It has been customary to speak of velocity in feet per minute, but feet per second is more convenient and is becoming more popular, especially in turbine work.

These curves are drawn for saturated steam and no correction curve for superheated steam is shown. Such a correction can readily be derived by taking the ratio of the corresponding specific volume, and proceeding as explained with the pressure-correction curves. It will be found that for every 50 degrees F. rise in superheat, the velocity will be increased 8 per cent. Referring to the above example, if the steam was superheated 100 degrees F. the velocity would have been 135×1.16 or 156.6 feet per second. Likewise, for every 1 per cent moisture in the steam, the velocity is decreased approximately 1 per cent.

Pressure-Drop Curve

Curve No. 3 shows the drop in pressure due to the flow of saturated steam in straight smooth pipes 100 feet in length at 100-pounds gauge pressure, and Curve No. 4 is a pressure-correction curve, so that the pressure loss for any condition of flow, pipe diameter, and static pressure can be obtained from these two curves.

Several formulæ have been derived for determining the pressure loss due to the flow of steam in straight smooth pipes; they were all employed in making the preliminary calculations, but the one most consistent with, and checking very closely, actual test results is the one here used and found on page 318 of Babcock & Wilcox's Handbook, "Steam," as follows,

$$p = 0.000131 \left(1 + \frac{3.6}{d}\right) \frac{w^2 L}{D d^5} \qquad \text{(III)}$$

or

$$w = 87 \left(\frac{p D d^5}{\left(1 + \frac{3.6}{d}\right) L} \right)^{\frac{1}{2}} \qquad \text{(IV)}$$

in which

w = pounds of steam flowing per minute
p = difference in pressure between the two ends of pipe in pounds per square inch

D = density of steam
d = internal diameter of pipe in inches
L = length of pipe in feet

It is readily seen that this equation is of the form $y = \sqrt{2\ px}$, or a parabola, and the resulting curve drawn with p as ordinates and w as abscissæ for any steam density (pressure) and pipe diameter will be a parabolic curve. Such a curve is difficult to draw and read accurately, whereas by taking the square root of the ordinate p and plotting the curve to a variable scale, a straight-line curve is produced and this inaccuracy eliminated. Curves Nos. 3 and 4 are therefore so drawn with a variable ordinate scale.

Curve No. 4 is a pressure-correction curve similar to Curve No. 2. From equation (**IV**) it will be seen that the drop in pressure p varies inversely with the density of the steam D, and as D varies with the pressure, the drop in pressure varies inversely with the static pressure of the steam. Hence by multiplying the ratio of the density of steam at 100-pounds gauge pressure to that at any other pressure by the drop in pressure at 100-pounds gauge pressure, the drop at the desired pressure is obtained, *i.e.*,

$$p_x = \frac{D_{100}}{D_x} \times p_{100} \qquad (V)$$

Following this principle, the pressure-correction Curve No. 4 is obtained for the various static pressures and the curves are straight lines when drawn to a square-root scale, as explained above. Likewise, the correction for superheat is derived, but as the density of steam is decreased by superheating, the pressure drop is increased above that shown by the curve for saturated steam and at the ratio of approximately 8 per cent for every 50 degrees F. increase in superheat. For every 1 per cent of moisture in the steam, the pressure drop is decreased approximately 1 per cent.

The loss in pressure as shown by Curves Nos. 3 and 4 is for straight pipes 100 feet in length, but for a given flow of steam, size of pipe, and pressure, the drop is proportional to the length, and if the pressure loss for any length of pipe other than 100 feet is desired, it is necessary only to divide the drop found from the curves by 100 and multiply by the length in question.

Elbows, globe valves and square-ended entrance to pipes all offer additional resistance

to the passage of steam. Many formulæ have been advanced for computing the length of pipe, in terms of the diameter, equivalent to such fittings that offer extra resistance, but they vary widely; and unfortunately those who have made actual experiments on the pressure loss in pipes under various conditions of flow, size of pipe, and pressure, seem to have failed to include such fittings in their experiments. It seems to be standard practice to allow for the extra resistance at the entrance of a pipe (where no special bell or gradual throat is provided) a length equal to 60 times the diameter of the pipe, for a right-angle elbow a length equal to 40 diameters, and for a globe valve a length equal to 60 diameters. The effect of gate valves, and bends, with a radius in feet equal to the diameter of the pipe in inches, is slight and can be considered the same as a straight section of pipe. Having obtained from Curves Nos. 3 and 4 the pressure drop per 100 feet for any given condition, the total drop can be obtained from any section of piping with valves, elbows, etc., by this equation:

$$p = \frac{p_1}{100} \times \left(L + \frac{60\ d}{12}E + \frac{40\ d}{12}El + \frac{60\ d}{12}Gv\right) \quad \textbf{(VI)}$$

in which

		Example
p	= total pressure drop	
p_1	= pressure drop per 100 feet	2 pounds
L	= length of pipe	50 feet
d	= diameter of pipe in inches	3 inches
E	= Number of entrances to pipe	1
El	= Number of elbows	3
Gv	= number of globe valves	2

Substituting the values shown in the example, we have

$$p = \frac{2}{100}\left(50+\frac{60\times3}{12}\times1+\frac{40\times3}{12}\times3+\frac{60\times3}{12}\times2\right)$$

$$= \frac{2}{100}(50+15+30+30) = \frac{2}{100}\times125 = 2.5 \text{ pounds, total pressure drop.}$$

It should be remembered that a pressure drop is not altogether a loss of energy, but a change into heat energy due to friction, and this energy is returned to the steam by reheating it, so that the only actual loss of energy is that due to the extra radiation from the pipe. The throttling calorimeter clearly demonstrates this point.

Therefore, the energy loss due to radiation is far more serious than pressure-drop loss. The evils of water of condensation in steam-piping systems are well known and precau-

tions are generally taken to guard against the effect, but not to such an extent the cause.

Radiation Losses

Heat loss by radiation is not governed by the quantity of steam flowing, as is pressure drop, but by the difference in temperature of steam and external air, size of pipe or square feet of radiating surface, the quality and thickness of pipe covering or coefficient of radiation, as shown in Table 1, page 16.

It will be noticed that the coefficient of radiation is given for lineal feet of pipe and not for square feet of pipe surface; this was done for convenience of comparison. It will be seen that the radiation loss is 29 per cent less for a 6-inch pipe covered with 1¼-inch covering than with ¾-inch covering; likewise, if a 4-inch pipe is used, instead of a 6-inch, both insulated with 1¼-inch covering, the 4-inch pipe will result in a saving in heat radiation of 27 per cent. Hence, it is of importance that the size of pipe employed be such that the combined heat losses, due to radiation and pressure drop, are a minimum. To this end Curve No. 5 is presented. In figuring this curve some standard quality and thickness of pipe covering, external tempera-

ture of air, and radiation per square foot of pipe area per degree difference in temperature between steam and external air have to be assumed. The following have been chosen as a fair average of standard conditions:

1¼-inch magnesia canvas-covered covering,

70 degrees F. temperature of air.

The radiation loss in B.t.u. per square foot pipe surface per hour per degree difference in temperature is shown in Table 2, page 17.

The radiation loss per hour per 100 feet of pipe was calculated according to the above conditions for the various pipe sizes and pressures, likewise the heat loss in B.t.u. per hour per 100 feet of pipe due to pressure drop. The drop in pressure corresponding to the various flows, pipe diameters, and pressures of steam considered was derived from Curves Nos. 3 and 4, and the heat loss due to this drop obtained from Marks & Davis Steam Tables. By comparing these losses, it was found that the most economical size of pipe, or the size resulting in the minimum heat loss, is obtained when the loss due to radiation is five times that due to loss in pressure. In a similar manner this comparison

Explanation of Curves Nos. 1, 2, 3 and 4

These curves will be found on a folded insert inside the back cover of this volume

Take for example,

20,000 pounds steam per hour
75 pounds steam pressure
6-inch pipe

To Find Velocity:—

At 20,000 pounds on Curve No. 1, read vertically downward to the 6-inch pipe diameter curve, then horizontally to the left; it will be seen that the velocity would be 107 feet per second if the pressure was 100 pounds gauge, but as the pressure is 75 pounds continue to the left to the 75-pound pressure line (Curve No. 2), then read vertically upward to the velocity axis, which shows a velocity of 135 feet per second, as indicated by the arrows.

To Find Drop in Pressure:—

Take for example,

30,000 pounds steam per hour
125 pounds steam pressure
7-inch pipe

At 30,000 pounds on Curve No. 3, read vertically upward to the 7-inch pipe diameter curve, then horizontally to the left; it will be seen that the drop in pressure would be between 1.00 and 1.21 pounds if the pressure was 100 pounds gauge, but as the pressure is 125 pounds, continue to the left to the 125-pound pressure line (Curve No. 4), then read downward to the drop in pressure axis, which shows

a drop in pressure between .81 and 1.00 pounds per 100 feet of pipe. The exact drop can be found by extracting the square root of the smaller scale number read, dividing the space between the two divisions into ten parts, and counting each small division as .01 and then squaring the result, so found, i.e.

$\sqrt{.81} = .9, + .06 = .96, \overline{.96^2} = .92$ pounds, drop in pressure.

Likewise if any three of the five variables, pressure, pipe diameter, flow, velocity and drop in pressure, are known the other two can be obtained.

For instance, take for example

Steam at 50-pounds pressure flowing at a velocity of 90 feet per second in a 10-inch pipe.

To find the pounds of steam flowing per hour and the resulting drop in pressure:—

At 90 feet velocity on Curve No. 2 read vertically downward to the 50-pound pressure line, then horizontally to the right to the 10-inch pipe line, Curve No. 1, then vertically upwards to the flow axis, showing a steam flow of 26,300 pounds per hour. To find the pressure drop, read vertically upward from this flow to the 10-inch pipe line Curve No. 3, then horizontally to the left to the 50-pound pressure line, then vertically downward to the pressure drop axis, showing a drop in pressure between .16 and .25 pound or .21 pound per 100 feet of pipe, i.e.,

$$\sqrt{.16} = .4, + .06 = .46, \overline{.46^2} = .21$$

Note:—

In reading Curves Nos. 2 and 4 do not cross horizontal axis between drop in pressure and velocity.

TABLE 1

B.T.U. LOSS PER LINEAL FOOT OF PIPE PER HOUR PER DEGREE DIFFERENCE IN TEMPERATURE OF STEAM AND EXTERNAL AIR *

Pipe Diam.	† Thickness of Pipe Covering, Inches					
	Bare	½	¾	1	1¼	1½
2	2.00	.478	.381	.323	.276	.255
4	3.5	.798	.607	.518	.45	.395
6	5.01	1.13	.865	.71	.615	.536
8	6.43	1.42	1.085	.89	.756	.67
10	7.98	1.77	1.34	1.09	.928	.81

was made for various static pressures and sizes of pipe and the Curve No. 5 § plotted to a square-root scale, which greatly smoothed out these curves. This curve shows the most economical size of pipe to be used, also the resulting drop in pressure per 100 feet of pipe; but as the pressure drop and radiation losses are proportional to the length and the extra radiation of the fittings makes up for their additional pressure loss, the curve holds for any length of pipe. To apply this curve, take for example 30,000 pounds steam per hour, 150-pounds gauge pressure, and 150 feet effective length of pipe; to find the most economical size of pipe, read vertically from the 30,000-pound flow line to the intersection of the 150-pound pressure line; this

* Calculated from data given on page 314, Babcock & Wilcox Steam Handbook.

† Covering magnesia, canvas-covered.

§ Curve No. 5 will be found as a folded insert inside the back cover of this volume.

also intersects the 6-inch pipe line, which shows that this size of pipe will result in the minimum energy losses. By reading horizontally to the left from this intersection, the pressure drop is found to be 1.88 pounds per 100 feet or 2.82 pounds for 150 feet of piping. If, for instance, the pressure had been taken at 125-pounds pressure, the pressure curve would have been intersected between the 6-inch and 7-inch pipe line, and the 7-inch pipe size used. To find the exact pressure drop under this condition, the Curves Nos. 3 and 4 would have to be referred to—*i.e.*, 30,000-pounds flow, 7-inch pipe, 125-pounds pressure, giving a pressure drop of .92 pounds per 100 feet or 1.38 pounds for 150 feet.

By comparing the values obtained from Curve No. 5 with the velocity Curves Nos. 1 and 2 it will be found that, under the assumed conditions of pipe covering and radiation, the most economical velocity to work

TABLE 2

RADIATION IN B.T.U. PER SQUARE FOOT OF PIPE SURFACE PER HOUR PER DEGREE DIFFERENCE IN TEMPERATURE OF STEAM AND EXTERNAL AIR, USING 1¼-INCH MAGNESIA COVERING

Pipe Diam. Inches	2	3	4	5	6	7	8	10	12
B.t.u.	.445	.405	.381	.265	.355	.345	.335	.329	.325

TABLE 3

STEAM VELOCITIES ADOPTED AS STANDARD BY VARIOUS MANUFACTURERS

Manufacturer	Velocity of Steam, Feet Per Second	Remarks
Boilers	60 to 70	At 100-pound gauge pressure, normal rating of boiler Boiler capable of 100 per cent overload
	45	Normal boiler rating saturated steam Boiler capable of 100 per cent overload
	75	Normal boiler rating, 100 to 150 degrees F. superheat Boiler capable of 100 per cent overload
Turbines	150	High-pressure admission pipe
	200	Low-pressure admission pipe
	400 to 600	Exhaust to condenser
Engines	20 to 40	Mean velocity in admission pipe, 100-pound gauge pressure Saturated steam
Summary of 10 builders	140 to 180	Average velocity in exhaust pipe Atmospheric exhaust

saturated-steam pipes of all sizes and pressures is 125 feet per second. With thinner pipe covering or superheated steam the velocity would be proportionately increased. This opens up a very interesting subject, for as previously mentioned, it is customary to speak of a pipe being worked at a certain steam velocity, not a certain pressure drop.

Since the introduction of steam turbines and large units, there has been a marked tendency towards higher steam velocities, though there is still a wide variation in the pipe velocities recommended by the different builders of steam machinery.

In preparing this volume, the author wrote to a number of the manufacturers on this subject and the information obtained appears in Table 3 opposite. These figures represent the manufacturers' standards and may vary widely according to the purchasers' specifications. It has been claimed that a high velocity in the boiler riser will cause the boiler to foam; this does not seem logical and has not proven so in practice. Foaming is more or less dependent upon the rate at which the boiler is driven and has no bearing on the velocity at which the steam is taken away, so long as the proper deflection plates and dry pipes are provided in front of the boiler outlet.

It will be noted that the reciprocating-engine builders employ a much lower velocity than the other builders, due to the fact that steam does not flow continuously, but intermittingly, in the supply pipe to an engine. Take for instance an engine operating at one-

fourth cut-off; the steam in the supply pipe is flowing theoretically only one-fourth of the time, so that the momentary velocity, when the engine is taking steam, is four times the mean velocity. Some engine builders claim satisfactory results are not obtained by placing at the engine a receiver of sufficient volume to supply the instantaneous quantity of steam demanded by the engine, and working the steam header at a higher velocity. The author has found this method very satisfactory, in some cases, not only in eliminating bad vibrations in the steam piping, due to the pulsating flow, but in producing an actual reduction in the drop in pressure, due to the elimination of the extra friction caused by the reciprocating flow of steam in the main header.

Though it is generally considered good practice to design steam piping, to be operated under continuous-flow conditions, for a velocity of 100 feet per second, it is extremely seldom this velocity is found in practice. In the majority of cases it will vary from 25 to 75 feet per second, and the author has found the velocity so low that a Pitot-tube deflection would hardly be noticed.

The designing engineer will invariably ac-

knowledge that the steam piping installed is entirely too large for the present needs, but that it was so designed to take care of future growth of the plant. But why plan only the piping for a 300 to 500 per cent future growth, when no allowance is made for such growth in other equipment? If first cost, repairs, and interest on this extra-size piping and the excessive heat losses due thereto are figured, it will often be found that the piping could have been replaced several times before the plant had grown to its pipe capacity. This point is well illustrated in a case where a belt line of 10-inch pipe of about 300 feet in length supplied an engine room that had been added to several times, until the total capacity was 500 horse power. The maximum steam demand was approximately 12,500 pounds per hour at 125-pounds pressure, resulting in a velocity of 10 feet per second in each leg; this is maximum conditions, remember. One leg of the 10-inch pipe was cut out and held in reserve and a 4-inch pipe substituted for the other, through which the velocity reached a maximum of 125 feet per second. When this was recommended, the operating engineer objected strenuously, saying that he would not be able to maintain steam

pressure in the engine room, because of the reduced pipe area, and that the additional coal required to carry a higher pressure to make up this pressure loss would be greatly in excess of the heat loss by radiation from the 10-inch pipe. Figuring this point out, the total heat in steam at 125-pounds gauge pressure is 1192.2 B.t.u. and assuming a pressure of 130 pounds had to be carried to make up for a 5-pound pressure loss, the total heat at 130-pounds gauge pressure is 1192.8 B.t.u. or .6 B.t.u. per pound of steam per hour must be added for an increase of 5 pounds pressure. If 12,500 pounds of steam is used per hour, 12,500 $\times$ .6 or 7,500 B.t.u. additional heat will be required per hour. Using coal of 12,500 B.t.u. per pound and a boiler efficiency of 60 per cent, there is available 7,500 B.t.u. per pound of coal, or it will require exactly 1 pound of coal per hour to carry the extra 5 pounds pressure. In the above-mentioned case there resulted no additional pressure drop by installing the 4-inch pipe, but to the contrary a considerable saving in the coal pile, largely due to shutting down two large ventilating fans, made necessary by the excessive radiation from the large-size pipes formerly used in the plant.

It is strange that the average operating engineer has such a fear of pressure-drop losses. They do not hesitate to make allowances for voltage drop in electrical transmission lines and design with care the proper

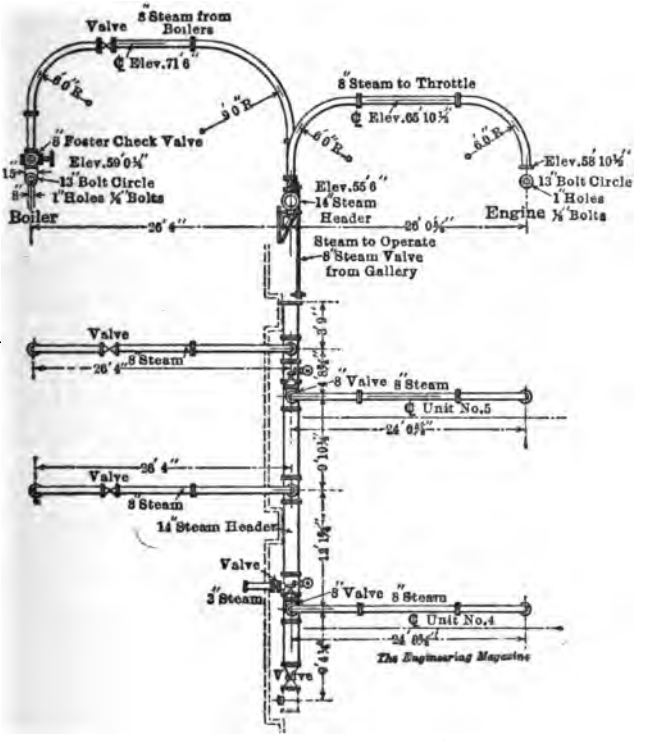


FIG. 1. TYPICAL FAULTY LAYOUT OF A CENTRAL-STATION PLANT

size of conductors to be used, but with the steam end it is different—doubtless due to the fact that electrical losses are so easily figured and are not governed by the numerous variables that enter into the calculation of steam data; strange, too, as it is from steam that the original cost of generating electric power is incurred.

The points to be brought out in this article may best be shown by taking some actual pipe installation and applying to it the curves here shown. Figure 1 is a piping layout in one of our moderate-sized central-station plants. The maximum flow in the 8-inch pipe from the boiler to the 14-inch header and to the engine is 26,000 pounds and the normal flow 17,000 pounds per hour at 150-pounds gauge pressure.

Length of 8-inch pipe from boiler to 14-inch header = 50 feet

Length of 8-inch pipe from header to engine = 37 feet

Total length, 87 feet

Effective length = $87 + \frac{60 \times 8}{12} \times 2 = 107$ feet + say 15 per cent for easy bends = 123 feet.

Velocity 26,000 pounds maximum flow, 8-inch pipe, 150-pounds pressure = 57 feet per second.

Velocity 17,000 pounds normal flow, 8-inch pipe, 150-pounds pressure = 37 feet per second.

Drop in pressure, 26,000 = pounds maximum flow per 100 feet = .30 pound.
Drop in pressure, 26,000 = pounds maximum flow per 123 feet = .37 pound.
Drop in pressure 17,000 pounds normal flow per 100 feet = .12 pound.
Drop in pressure 17,000 pounds normal flow per 123 feet = .15 pound.

From Curve No. 5 it will be found that for 26,000-pounds steam flow at 150-pounds pressure, the most economical size of pipe is between 5 and 6 inches, and if a 6-inch pipe is used the above values would become:

Velocity 26,000-pounds maximum flow, 6-inch pipe, 150-pounds pressure = 98 feet per second.
Velocity 17,000-pounds normal flow, 6-inch pipe, 150-pounds pressure = 64 feet per second.
Drop in pressure 26,000 pounds maximum flow per 100 feet = 1.37 pounds.
Drop in pressure 26,000 pounds maximum flow per 123 feet = 1.68 pounds.
Drop in pressure 17,000 pounds normal flow per 100 feet = .58 pound.
Drop in pressure 17,000 pounds normal flow per 123 feet = .71 pound.

It will be seen that the use of a 6-inch pipe does not result in excessive steam velocities nor pressure drop, and by providing a receiver of sufficient volume at the engine better results could be obtained than with the present 8-inch piping, without a receiver, as the

14-inch header is not close enough to the engine to serve this purpose. This is by no means an extreme case; many worse ones could be cited.

The cost of material, labor and covering for an installation of 8-inch high-pressure piping is about 25 per cent more expensive than a similar installation of 6-inch piping, and this saving in favor of the smaller pipe is increased by the reduction in heat losses and repair charges. In cases where pipes two to three sizes too large are used, the first cost is proportionally increased, and it is hard to mention any other items in the design of a power plant where from 25 to 50 per cent unnecessary increase in the first cost is so often found as in the piping installation.

Of course, every installation of piping has to be designed according to the conditions to be met with in practice, and no definite rule as to the exact size of pipe, thickness of covering, etc., can be made. In one case it may be advisable to design the piping for a velocity of 150 feet per second, while in another, half this velocity may be found most economical. It is hoped this chapter and the curves here shown will be of service to the designing engineer in deciding this point.

Chapter II

SPECIAL CONDITIONS AFFECTING LOW-PRESSURE SYSTEMS

BEFORE designing the piping layout for a steam-heating system, it is necessary to determine the amount of heating or square feet of radiating surface required, and the source of supplying the steam; then the distribution system—the piping—connecting the source of supply with the point of use is considered, and the latter item will be treated in this chapter.

No attempt will be made to discuss the various methods of calculating the square feet of radiating surface required for different types of buildings, nor the merits of the various methods of steam heating, namely, direct, indirect, semi-indirect, employing the numerous wet- and dry-pipe return systems on the market. In fact, there seems to be such a diversity of opinion as to the value of the various systems of steam heating in use, that an article on the subject is only an expression of one's personal views in some

particular case. This fact is not due to any lack of co-operation on the part of the engineering profession, but due to each installation presenting a problem within itself that has to be solved to meet the conditions under which it is to be operated.

It is my endeavor to present certain fundamental principles that have been somewhat generally accepted, in such form as to be useful to the engineer in the general design of the piping layout for steam-heating systems and other installations using low-pressure steam.

The item of first importance in designing a heating system is that the radiating surface be ample. No system of circulation can do more than keep the radiating surface at the temperature of the steam, and only a small variation in temperature can be judiciously obtained by varying the steam pressure. Next in importance is the circulating system, for without the proper size of steam mains and connections the radiating surface is unable to perform its duty. There is a continual loss in the temperature and pressure of steam as it leaves its source of supply, due to radiation and friction pressure losses. These losses may be kept to a minimum by

properly proportioning the size of pipe used, and result in a saving in both the installation and operating costs.

The energy loss due to radiation is not affected by the quantity of steam flowing in a pipe, whereas the resulting drop in pressure varies as the square of the flow of steam; that is, by doubling the flow the pressure drop is increased four times; therefore, in determining the size of mains, the resulting drop in pressure is the governing feature, and a drop-in-pressure curve is shown (Curves 8 and 9).*

As the drop in pressure is directly dependent upon the quantity or velocity of steam flowing, as previously stated, and as it is easier to calculate velocity than pressure drop, velocity therefore being more generally used, a steam velocity curve is also presented (Curves 6 and 7).

Coefficient of Condensation

To determine the velocity of steam flowing in a pipe, the quantity of flow has to be known, and as it is customary to rate heat-

* Curves 6, 6A, 7, 8, 8A, and 9 will be found on a sheet folded and inserted within the back cover of this volume.

ing systems in square feet of radiating surface, it is necessary to assume some value for the coefficient of condensation per square foot of radiation. This coefficient will vary with conditions of use; but as long as a standard is chosen and all calculations are based on it, any value can be taken, and if any other coefficient is preferred it can be applied by direct proportion. It has been found in practice that under ordinary conditions of direct radiation, with steam at atmospheric pressure and 0 degrees F. outside temperature, each square foot of radiation surface will condense 0.3 pounds of steam per hour. For indirect radiation this figure will be increased three or four times, but for direct radiation it can safely be taken as a fair maximum under ordinary operating conditions, and is here used, though the curves, 6, 7, 8, are drawn with a double scale—square feet of radiating surface and pounds of steam per hour —so that the curves can be used for any other value of the coefficient of condensation by simply multiplying the square foot of radiation by the coefficient chosen to obtain the maximum pounds of steam condensed per hour, and reading from the value so found on the pounds-flow scale.

Derivation of Curves

In calculating the steam-velocity and drop-in-pressure curves here shown, the same formulæ and methods are employed as in the preceding chapter and will be only briefly explained.

Steam-Velocity Curve

Curve 6 is drawn between square feet of radiation surface or steam-flow pounds per hour and the resulting velocity for the various standard sizes of pipes at 14.7 pounds absolute pressure, and calculated from the formula

$$V = \frac{F\,S}{25\,A} \qquad \text{(VII)}$$

in which

V = velocity of steam in feet per second.
F = flow of steam in pounds per hour.
S = specific volume of steam at 14.7 pounds absolute pressure.
A = internal area of pipe in square inches.
25 = constant resulting by reducing hours to seconds and square feet to square inches.

By substituting in this equation for any chosen flow at 14.7-pounds absolute pressure, the velocities for the various sizes of pipes

are obtained, and if the velocity at any other pressure is desired it can be found by referring to the pressure-correction lines, Curve 7, which are derived by taking the ratio of the specific volume of the pressure in question to that at 14.7 pounds pressure and multiplying by the velocity at the latter pressure, *i.e.*,

$$\text{Velocity at pressure X or } V_x = \frac{Sx}{S_{14.7}} \times V_{14.7} \quad \textbf{(VIII)}$$

To use these curves, suppose, for example, it is desired to find the velocity of steam flowing in a 7-inch pipe supplying 12,000 square feet of radiating surface condensing a maximum of 3,600 pounds of steam per hour (12,000 × .3) at 20-pounds absolute pressure. Referring to the horizontal axis of Curve 6 at 12,000 square feet or 3,600-pounds flow read vertically downward until the 7-inch pipe diameter curve is intersected, then to the left until the 20-pound line on Curve 7 is reached, then vertically upward to the velocity axis showing a velocity of 74 feet per second. Likewise if the velocity is known for any pressure and size of pipe, the pounds flow per hour can be found by reversing the above process. Curve 6-A is an enlargement of

the lower section of Curve 6, to insure accurate reading. These curves are drawn for saturated steam, but the corrected velocity for any steam condition can be obtained by deducting 1 per cent for every per cent of moisture in the steam, and adding 8 per cent for every 50 degrees of superheat.

Drop-in-Pressure Curve

The formula used in calculating the drop in pressure, due to friction as in the previous chapter, is

$$p = 0.000131 \left(1 + \frac{3.6}{d}\right) \frac{w^2 L}{D\, d^5} \quad \text{(IX)}$$

or

$$w = 87 \left(\frac{p\, D\, d^5}{\left(1 + \frac{3.6}{d}\right) L} \right)^{\frac{1}{2}} \quad \text{(X)}$$

in which

w = pounds steam flowing per minute.
p = difference in pressure between the two ends of pipe in pounds per square inch.
D = density of steam.
d = internal diameter of pipe in inches.
L = length of pipe in feet.

As the drop in pressure so obtained varies as the square of the flow (or square feet of ra-

diating surface) the resultant curve is a parabola; but by plotting the pressure-drop ordinate to a variable scale (square root) the parabolic curve becomes a straight line and more accurately plotted. Curves 8 and 8-A are so drawn, curve 8-A being the same as a portion of Curve 8 drawn on a larger scale. These curves give the pressure drop in ounces, per 100 feet of straight smooth pipe, for various sizes of pipe, when worked at atmospheric pressure. The drop in pressure for any other static pressure can be found by referring to Curve 9, as previously explained, for determining the velocity. Take for example the above mentioned case, *i.e.*, 12,000 square feet radiating surface or 3,600 pounds steam per hour flowing in a 7-inch pipe at 20 pounds absolute pressure; having found the velocity to be 74 feet per second, to determine the drop in pressure per 100 feet of straight smooth pipe: at 12,000 square feet or 3,600-pounds flow on the horizontal axis of Curve 8 read vertically upwards until the 7-inch pipe diameter curve is intersected, then horizontally to the left to the 20-pound pressure line of Curve 9, then read vertically downward to the pressure-drop axis, which shows a drop between 1.21 and 1.44 or 1.30 ounces per 100

feet of pipe. In a like manner if any three of the five variables, *i.e.*, pipe diameter, flow of steam or square feet radiating surface, static pressure, drop in pressure, and velocity, are known the two unknown quantities can be obtained from these curves. For instance, say the velocity in a 7-inch pipe at 25-pounds absolute pressure is 60 feet per second; to find the resulting steam flow and pressure drop: at 60 on the velocity axis Curve 7, read vertically downward until the 25-pound pressure curve is intersected, then horizontally to the right to the 7-inch pipe-diameter line, then vertically upward to the flow or square feet radiating surface axis, which gives 11,900 square feet radiating surface or 3,570 pounds flow. Now to find the drop in pressure, proceed from this point upward (Curve 8) until the 7-inch pipe-diameter curve is reached, then horizontally to the left until the 25-pound pressure line is intersected (Curve 9), then downward to the drop-in-pressure axis, showing a drop in pressure of 1.04 ounces per 100 feet of pipe.

The drop in pressure is proportional to the length of pipe; the additional loss, due to globe valves, bends and elbows, can be allowed for by treating each fitting as an addi-

tional length of pipe, and a convenient formula for calculating the equivalent length is:

$$* \quad A = \frac{114d^2}{d + 3.6}; \; B = 2/3 \; A \qquad \textbf{(XI)}$$

in which

d = Diameter of pipe in inches.
A = Length, in inches, of pipe equivalent to globe valve.
B = Length, in inches, of pipe equivalent to bends and elbows.

From this equation the following values are obtained:

TABLE 4

Pipe Diameter	Equivalent Straight Pipe Due to Globe Valves		Bends and Elbows	
	Feet	Inches	Feet	Inches
1	2	1	1	5
1½.....	4	2	2	10
2	6	9	4	6
2½.....	9	9	6	6
3	12	11	8	8
3½.....	16	5	10	11
4	20	0	13	4
4½.....	23	9	15	10
5	27	7	18	5
6	35	8	23	9
7	44	0	29	4
8	52	5	34	11
9	61	1	40	9
10	69	10	46	7
12	87	8	58	6

* Taken from Ingersoll-Sergeant Drill Co.'s Catalogue.

Thus in the preceding example, if the total length of pipe was 500 feet with four globe valves and ten elbows, the total effective length would be 500 feet + 4 × 44.0 feet + 10 × 29.34 feet, or 969 feet (see Table 4 for 7-inch pipe), and the total drop in pressure would be 1.04 oz. × (969 ÷ 100) feet, = 10. ounces or .63 pounds, resulting in an effective steam pressure at the point of use of 25 — .63 or 24.37 pounds absolute.

Allowable Drop in Pressure

Now the question arises, what is the most economical drop in pressure permissible for low-pressure heating systems? This, of course, depends upon conditions of use. Where there are exceptionally large and lengthy runs, or where the system covers a number of scattered buildings, each building should be considered as a separate problem; but under ordinary conditions the drop in pressure should not exceed 1 ounce per 100 feet of straight pipe. This maximum drop can be used with very satisfactory results for all steam mains, risers, and radiator connections in buildings of ordinary size, but where low-pressure steam is distributed over

large areas a lower drop is often necessary, and where the distribution is made by high-pressure and reduced at the point of use, a higher drop should be allowed in the high-pressure system, as shown by Curve 5 in the preceding chapter. Therefore, adopting as a standard a drop in pressure of 1 ounce per 100 feet of pipe for low-pressure heating systems, and reading from Curves 6, 7, 8, and 9, at this drop, the corresponding values of flow and velocity for the various pipe diameters and pressures, and plotting the values so found, Curve 10 is produced.* This curve is convenient in determining the economical size of pipe to be used for any quantity of radiation at several different pressures (plotted for convenience as pounds *gauge* pressure); it also shows the resulting velocity. For instance, take a system containing 10,000 square feet of radiating surface supplied by steam at 5-pounds gauge pressure, it being desired to know the size of the supply main that will result in a loss in pressure of 1 ounce per 100 feet of pipe. Referring to Curve 10 at 10,000 square feet, read vertically upward until the 5-pound pressure line is intersected;

* Curves 10 and 10A will be found on a separate sheet, folded and inserted within the back cover of this volume.

EXPLANATION OF CURVES

Nos. 6 and 6-A, 7, 8 and 8-A, and 9

Example:—

A 7-inch steam main supplying 12,000 square feet direct radiation, or 3,600 pounds steam per hour, at a maximum pressure of 20 pounds absolute.

To find resulting velocity of steam.

At 12,000 square feet radiation or 3,600 pounds steam per hour (12,000 × 0.3) on Curve No. 6 read vertically downwards to the 7-inch pipe-diameter line, then to the left; it will be seen that the velocity would be 99 feet per second if the pressure was 14.7 pounds absolute, but as it is 20 pounds, continue to the 20-pound line of the pressure-correction Curve No. 7, and at this point read vertically upwards to the horizontal velocity axis, showing a velocity of 74 feet per second, as per arrow indications.

To find resultant drop in pressure.

At 12,000 square feet or 3,600 pounds steam per hour, on Curve No. 8, read vertically upwards to the 7-inch pipe-diameter line, then horizontally to the left to the 20-pound absolute pressure line Curve No. 9, then vertically downward to the horizontal drop in pressure axis, showing a drop in pressure between 1.21 and 1.44 ounces per 100 feet of straight smooth pipe. As it is a variable ordinate scale (square root) the exact reading can be obtained by extracting the square root of the smaller scale number read and dividing the space between the two divisions into ten parts and adding each small division as .01 and then squaring the result so found, i. e.,

$$\sqrt{1.21} = 1.1$$

$$1.1 + .04 = 1.14$$

$$1.14 = 1.30 \text{ oz. drop per 100 feet}$$

Likewise, if any three of the five variables, namely static pressure, pipe diameter, velocity, square feet radiation or pound steam flow and drop in pressure, are known the other two can be obtained. The method of using the Curves for this purpose is illustrated as follows:

Example:—

Assuming a velocity of 85 feet per second at 18-pound absolute pressure, 9-inch pipe.

To find the resulting flow of steam or square feet of direct radiation and the drop in pressure.

At 85 on the horizontal velocity axis of Curve No. 7, read vertically downward to the 18-pound absolute-pressure line, then horizontally to the right to the 9-inch pipe-diameter line Curve No. 6, then vertically upward to the flow and square foot radiation axis, showing a flow of 6,000 pounds per hour and a radiation of 20,000 square feet. From this point read vertically upward to the 9-inch pipe diameter line Curve No. 8, then horizontally to the left to the 18-pound pressure line Curve No. 9, then vertically downward to the horizontal pressure-drop axis, showing a drop in pressure per 100 feet of pipe between 1.00 and 1.21 or 1.13 oz., i. e.

$$\sqrt{1.0} = 1$$

$$1 + .065 = 1.065$$

$$1.065^{8} = 1.13 \text{ oz.}$$

Note.

In using Curves do not cross horizontal axis between drop in pressure and velocity of Curves Nos. 7 and 9. Read Curves Nos. 6-A and 8-A same as Curves Nos. 6 and 8.

EXPLANATION OF USE OF CURVE NO. 10

Example:—

A heating system containing 10,000 square feet (3,000 pounds steam per hour) direct radiating surface steam pressure 5-pound gauge.

To find size of supply main resulting in a drop in pressure of 1 oz. per 100 feet of straight smooth pipe.

At 10,000 square feet radiating surface on the horizontal axis, read vertically upward to the 5-pound pressure line; this line is intersected between the 6-inch and 7-inch pipe-diameter line, thus a 7-inch pipe would be the required size. By reading from this point to the velocity axis at the left, the approximate velocity is found to be 65 feet per second, as per arrow indications. By referring to Curves Nos. 6 and 7 the exact velocity is found to be 62 ft. per second.

To find size of return mains resulting in a drop in pressure of 1 ounce per 100 feet of straight smooth pipe.

Divide the square feet direct radiating surface by the constant in the following table corresponding to the per cent of steam in returns and refer the radiation so found to Curve No. 10 or 10-A at atmospheric pressure.

Wet Return	2½% Steam	5% Steam	7½% Steam	10% Steam	15% Steam	20% Steam	25% Steam
+ 40	+ 20	+13.33	+ 10	+ 8	+ 5.71	+ 4.44	+ 3.64

Example:—

10,000 square feet direct radiation

10 per cent steam required to maintain dry returns

From table K = 8 and equivalent radiation $= \frac{10,000}{8}$

= 1,250 square feet

Referring to Curve No. 10 or 10-A at 1,250 square feet radiating surface and reading upward to the atmospheric pressure line, it is intersected between the 3-inch and 3½-inch pipe line; therefore 3½-inch pipe is the most economical size to use.

this occurs between the 6-inch and 7-inch pipe lines; therefore a 7-inch pipe would be the proper size to use, and by reading to the left at the intersection of the pressure line the velocity is shown to be 65 feet per second, though a 7-inch pipe supplying 10,000 square feet will result in a slightly lower velocity. If the exact velocity is desired, it can be found by referring to Curves 7 and 8, and will prove to be 62 feet per second. In like manner the proper sizes of the various branch risers can be obtained. Curve 10-A is the lower section of Curve 10 drawn to a larger scale to facilitate reading.

Return Mains

It has been customary to consider return mains as bearing a certain proportionate size to the supply mains, but this relation varies widely with the system of heating employed. The drop in pressure in the return mains should be the same as in the supply piping, namely 1 ounce per 100 feet of straight pipe, and the corresponding rating will be proportional to the square root of the ratio of the density of the steam supplied and that of the returned fluid (water and vapor). As water

at 212 degrees F. is 1,600 times heavier than steam at atmospheric pressure, a wet return pipe will handle 40 (square root of 1,600) times as much radiation as a steam-supply pipe with the same friction lost in pressure. For instance, 10,000 square feet radiating surface at 5-pounds gauge pressure requiring a 7-inch supply pipe will require a return pipe sufficient to supply 10,000 ÷ 40 or 250 square feet of radiating surface at atmospheric pressure, or a 2-inch return pipe (read from Curve 10-A at 250 square feet on atmospheric pressure line). The return pipes for dry return systems must not only be large enough to carry the condensed steam, but also the steam necessary to supply the heat lost by the returns. If in the above case it is assumed that 10 per cent of the steam delivered to the radiators is required to maintain the dry return system, the required size of the return main would be 3½ inches, *i.e.*, 10,000 square feet radiating surface, of which the return system has to be capable of handling 10 per cent steam or 1,000 square feet radiating surface, also water at 212 degrees F., equivalent to 10,000 ÷ 40, or 250 square feet, resulting in a total equivalent of 1,000 + 250 or 1,250 square

feet radiating surface. From Curve 10-A, we find that 1,250 square feet intersects the atmospheric pressure line between the 3-inch and 3½-inch pipe line; therefore the 3½-inch pipe would be the proper size to use.

For any per cent of steam in returns, the size of return pipe, for a friction loss of 1 ounce per 100 feet of pipe, can be found by dividing the corresponding constant in the following table by the square feet of radiat-

TABLE 5

TABLE OF CONSTANTS FOR DETERMINING SIZE OF RETURN PIPES

Wet Return	2½% Steam	5% Steam	7½% Steam	10% Steam	15% Steam	20% Steam	25% Steam
+ 40	+ 20	+13.33	+ 10	+ 8	+ 5.71	+ 4.44	+ 3.44

ing surface supplied and applying the equivalent square feet of radiation so found to Curve 10 or 10-A, using the atmospheric-pressure line.

By covering the return pipes, somewhat smaller sizes can be used than are otherwise suitable, as the heat loss will be reduced, necessitating less steam to be carried; likewise, vacuum return systems, the temperature being lower, require considerably less steam,

and a proportionate reduction in the size of return pipes can be made, the amount depending upon existing conditions.

These curves and those contained in the preceding chapter conclude the discussion of the laws governing the flow of steam and the economical design of steam piping. In the following chapter the use of exhaust steam for heating purposes, and effect of back pressure on engine efficiency, will be treated.

Chapter III

SAVINGS OBTAINABLE FROM EXHAUST STEAM

IT is a strange fact that the average large user of heat and power has failed to inform himself regarding the economical operation of this department of his business. Every other department may receive his close study, but the heat and power department, so long as it does its allotted duty, is not given much consideration, and it is not uncommon to find industrial plants where a saving of from 25 to 75 per cent could be effected. Many will acknowledge that this inefficiency exists, but will excuse themselves by saying, "It is one of the things that cannot be helped, and as our heat and power department cost is only a small per cent of the total production cost, say 5 per cent, a saving of, say, 20 per cent in this department is only a one per cent total saving—a small item." But why should not a saving in one department be sought after as much as in another; the owner would not hesitate to avail himself

of an equivalent saving in time or material used in the production of his article of manufacture; whereas with the heat and power department it is different. And why? Simply because the heat and power department, in the majority of cases, is not directly a producer and receives no credit for the work it does for the various other departments. The time and material furnished each department is charged up against that department and it is expected to deliver a certain quantity of finished goods, but no charge is made for the heat and power used. If one department is able to produce more goods it is given credit for its increased efficiency, but the power department has no credit side on the factory ledger, only a debit side, and simply has to answer the demands of the various departments over which it has no control, and not until it is unable to supply these demands is it given any consideration.

It will be understood, of course, that I am referring to large users of heat and power (factories and large public buildings) especially those using steam for heating, cooking, and drying purposes.

The owner is not altogether to blame for his power-plant inefficiency, as the manufac-

ture and transmission of steam is an invisible process—in boilers and pipes—and the lack of reliable means for measuring the flow of steam is responsible for the habit of neglecting this department.

Ever since the introduction of electricity we have had and made free use of electrical meters, but it has only been within the past few years that reliable steam meters have been on the market, and these meters, with the accompanying coal and water weighers, have done more to demonstrate the wasteful use of steam and arouse the interest of the plant owner than anything I know of. In one large plant with which I was connected, the newly installed steam meters unearthed leaks and other waste amounting to over $60,000 a year, and this is by no means an extreme case. I could quote many proportionately worse, but simply mention this to show the field for saving that even the largest plants have overlooked.

Since the introduction of the high efficiency steam turbine and of the internal combustion engine it has been stated that the steam engine will be wiped from the industrial field as completely as the American buffalo has from the prairies of the West. This is undoubted-

ly true of the large condensing engines used for central-station purposes, but where there is any demand for exhaust steam for heating, cooking, or drying purposes the simple steam engine will maintain its supremacy and become more and more used as the value of exhaust steam for this purpose is realized.

The engine builder, realizing the necessity of maintaining a demand for his product, has done much in the form of advertising to bring before the public the value of exhaust steam.

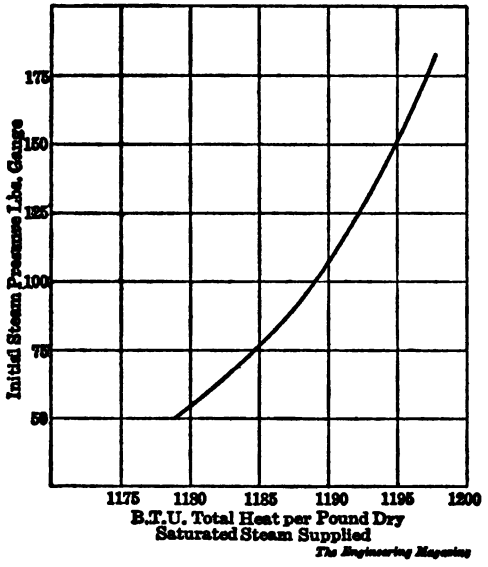
The best thermal efficiency claimed for a steam turbine is about 25 per cent and for an internal combustion engine about 30 per cent, whereas by using the exhaust steam from an engine for heating purposes a thermal efficiency from 75 to 90 per cent is not uncommon in practice. The theoretical perfect efficiency of a turbine operating at 200-pounds gauge pressure, 200 degrees F. superheat, and 29-inch vacuum is only 32.4 per cent, so that the limit has been very closely approached and there is no possibility of the steam turbine approaching the efficiency of the by-product steam engine.

The above statement can be more forcibly impressed by considering the heat required to convert one pound of water at 32 degrees

F. into steam at, say, 115 pounds absolute pressure as follows:

(1)	Heat required to raise one pound water at 32 degrees to water at 212 degrees F.	= 180	B.t.u.of sensible heat
(2)	Heat required to convert one pound water at 212 degrees to steam at 212 degrees	= 970.3	B.t.u.of latent heat
(3)	Heat required to raise one pound steam from 212 degrees to 338.1 degrees, the temperature of steam at 115 pounds absolute pressure	= 38.5	B.t.u.of sensible heat
	Total	1188.8	B.t.u.

It requires the same heat from the furnace to produce one B.t.u. of latent heat of evaporation as it does one B.t.u. of sensible heat, but there is yet to be devised a method of converting this enormous quantity of latent heat into mechanical energy, and, unless this heat is used for heating, cooking, or boiling purposes, this energy in the steam is wasted—except where it is partly used for heating the feed water, and then only about one-seventh



CURVE No. 11

TABLE 6

PERCENTAGE OF TOTAL HEAT REMAINING IN EXHAUST STEAM AT VARIOUS STEAM CONSUMPTIONS

Pounds Steam Consumed by Non-condensing Engine per I.h.p.	Per Cent of Total Heat B.t.u. Supplied Contained in Exhaust Mixture
20 to 25	89
25 to 30	91
30 to 35	92.5
35 to 40	93.5
40 to 45	94.5
45 to 50	95
50 to 60	96
60 to 70	96.5
70 to 80	97
80 to 100	97.5

of it is used, as generally the feed water enters at 70 degrees F. and is raised to about 210 degrees F.

A practical case will best show the point in question. Take a simple automatic cut-off governor engine operating at 100 pounds gauge pressure, 5 pounds back pressure, requiring 40 pounds of dry saturated steam per i.h.p. The thermal efficiency of this engine will be $2{,}545 \div (1{,}189 \times 40) = 5.3$ per cent, and there remains in the exhaust steam mixture 94.7 per cent of the total heat supplied; but not all of this is available for heating purposes, as part of the mixture contains water of condensation, amounting to 3 per cent in this case, so 97 per cent of the heat of the exhaust mixture is available as dry steam. As the total heat of steam at the exhaust pressure of 5 pounds gauge is 1,156 B.t.u., there is available 97 per cent of 1,156, or 1,121 B.t.u. per pound of steam, and this is $1{,}121 \times 100 \div 1{,}189$, or 94.3 per cent of the total heat supplied. Curve No. 11 and accompanying table (No. 6) are presented for obtaining this value quickly. Referring to the table and curve for example, the total heat of steam at 100 pounds gauge equals 1,189, and the percentage of total heat available in the exhaust, corresponding to 40 pounds steam consumption per

i.h.p., is 94.5 per cent—which checks the above (94.3 per cent) sufficiently close for practical purposes, and as close as the engine consumption will be known.

Again, referring to the useful heat in the steam—1,121 B.t.u.—if it is used for heating purposes and condensed to water at 212 degrees, 1,121 — 212 or 909 B.t.u. will have been used for this purpose, or $909 \times 100 \div 1{,}189 = 76.6$ per cent of the total heat supplied is used for heating. There is also a credit to be given for the heat returned to the boiler in the feed water, as we started with water at 32 degrees. This is 212 — 32 = 180 degrees or 15.1 per cent. Summarizing the total energy extracted from the original steam supplied the engine, we have 5.3 per cent converted into mechanical energy, 76.6 per cent used for heating purposes, and 15.1 per cent returned to the boiler in feed water, or a total thermal efficiency of 97 per cent. This does not take into consideration any engine or pipe heat losses, which will be governed by the length and size of pipe and covering used, but it will be seen that an over-all thermal efficiency in practice between 75 and 90 per cent is not impossible, nor is it uncommon in a well designed plant.

It will also be noted that the power developed by the engine is nothing more than a by-product and the exhaust steam from it is really the useful product.

It may prove interesting to figure out the loss in tons of coal if the exhaust steam were wasted in the above case and live steam used instead for heating. This is not infrequently the case; I have just inspected a plant where the exhaust from four pumps could be used for heating water by running a pipe not over 10 feet, and the live steam now used would be saved. Say the engine is 100 horsepower using 40 pounds of steam per horsepower, as in the above example, operating 10 hours per day and 300 days per year; there will be exhausted 12,000,000 pounds of steam containing 1,121 B.t.u. per pound, or a total of 13,452 million B.t.u. If coal costing $3.00 per ton, having a calorific value of 12,500 B.t.u. per pound, is used and a boiler efficiency of 60 per cent (a fair operating condition) each pound of coal will produce 7,500 B.t.u. of steam, thereby requiring 1,800,000 pounds of coal, or 900 tons, or $2,700 per year wasted. Does such a condition exist in your plant?

Where a temperature not in excess of the

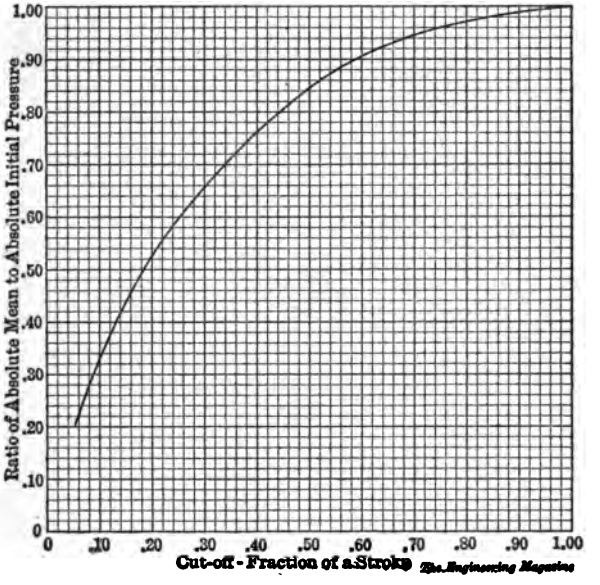
exhaust-steam temperature is required, exhaust steam is just as good as live steam for heating purposes—even better, as the gentle puffs of the engine exhaust seem to aid the circulation of the steam through the heating system and prevent air pockets being formed.

Effect of Back Pressure on Engine Efficiency

With a reasonable amount of back pressure on an engine the effect is not so serious as generally believed. Naturally if the back pressure is increased and other conditions unchanged, the power developed by the engine will be reduced; but this can be overcome either by increasing the length of cut-off, thereby requiring a greater amount of steam, or by increasing the initial steam pressure, or both. The most advisable method depends upon operating conditions. If there is a demand for the increased exhaust steam it is better to increase the cut-off, but if the extra exhaust steam cannot be used it is more economical to increase the initial pressure.

An example will best explain these points: Take a non-condensing engine, operating at 115 pounds absolute initial pressure, 15

pounds absolute exhaust pressure, and one-quarter cut-off at full load, where it is desired to use the exhaust steam for heating



CURVE No. 12

purposes at 20 pounds back pressure. Required to find:

1. The reduction in power developed by the engine due to the increased back pressure, the initial pressure and cut-off remaining the

same. The output of the engine under these conditions is proportional to the mean effective pressure in the two cases, which is obtained from the familiar equation:

$$Pm = p_1 \frac{1 + \text{hyp. log } R}{R}$$

in which

Pm = absolute mean pressure
p_1 = absolute initial pressure
R = ratio of expansion

For convenience, Curve No. 12, derived from this equation, is used.

At one-quarter cut-off the ratio of absolute mean to absolute initial pressure is 0.596 (read from curve), and, as the initial pressure is 115 pounds absolute, the theoretic mean is 115 × 0.596, or 68.54 pounds absolute. By increasing the back pressure 5 pounds, the initial pressure and cut-off remaining the same, the mean pressure becomes 68.54 — 5, or 63.54 pounds absolute; and as the output of the engine is proportional to the mean pressure, the reduction in power will be (68.54 — 63.54) × 100 ÷ 68.54 or 7.3 per cent. Thus, by increasing the back pressure on the engine 5 pounds, the output of the engine is decreased 7.3 per cent.

2. To find the increase in steam consumption due to increasing the back pressure 5 pounds, the initial pressure and power developed remaining the same. For the engine to do the same amount of work, under this condition, the same mean pressure must exist; therefore, a greater amount of steam has to be supplied by increasing the cut-off, and the problem resolves itself into finding the cut-off under the new conditions. With an absolute mean pressure of 68.54 pounds absolute and an increased back pressure of 5 pounds, the mean pressure for the engine to do the same work becomes 68.54 + 5 or 73.54 pounds, and the ratio of absolute mean to absolute initial pressure, under the new condition, is 73.54 ÷ 115 or 0.64; and this ratio, from the curve, gives a cut-off of 0.28. As the steam consumption of an engine is proportional to the cut-off the resulting increase in steam consumption becomes (0.28 — 0.25) × 100 ÷ 0.25 or 12 per cent.

3. To find the necessary increase in initial pressure due to a 5-pound increase in back pressure, the cut-off and power output remaining the same. At one-quarter cut-off the ratio of absolute mean to absolute initial pressure is 0.596, and, as above, the absolute

mean pressure is 73.54 at 5 pounds back pressure; therefore, the initial pressure becomes 73.54 ÷ 0.596 or 123.4 pounds absolute, or an increase in pressure of 8.4 pounds. This increase in pressure also causes a slight increase in steam consumption due to the increase in steam density. The density of steam at 115 pounds absolute is 0.2577 pounds per cubic foot, and at 123.4 pounds it is 0.2756 pounds. As the point of cut-off is assumed to be the same in both cases, the same volume of steam will be supplied; but in the latter case 0.2756 — 0.2577 or 0.0179 pounds per cubic foot, or 6.9 per cent more in weight is required.

One pound of steam at 115 pounds absolute contains 1,188.8 B.t.u., and at 123.4 pounds, 1,190 B.t.u.; hence, each pound of steam raised to the higher pressure requires 1.2 B.t.u. more per pound of steam, or one-tenth of one per cent. Therefore, under the third condition, 106.9 per cent × 100.1 per cent — 100 per cent, or 7 per cent more heat, and consequently that much more coal, will be required.

Summarizing the above we have, by increasing the back pressure 5 pounds, the choice of three effects:

1. Maintaining the same initial pressure and steam consumption with a 7.3 per cent loss in power output of the engine.
2. Maintaining the same initial pressure and power output of the engine by increasing the steam-coal-consumption 12 per cent.
3. Maintaining the same cut off and power output by increasing the initial pressure 8.4 pounds and steam consumption 7 per cent.

These conditions will, of course, vary somewhat in actual practice according to the type and condition of the engine used, but the comparison conclusively shows the advisability of sacrificing a small efficiency in the operation of the engine to gain the manifold increased efficiency of the plant as a whole.

From these facts it is hard to conceive why a factory or building of any size, requiring heat and power, does not take advantage of the wonderful over-all efficiency offered by the simple steam engine; and if our business men will allow themselves to be educated regarding their heat and power requirements as they have their other departments, the old fa-